NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 555

AIR FLOW AROUND FINNED CYLINDERS

By M. J. BREVOORT and VERN G. ROLLIN

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AERONAUTIC SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Abbreviation	Unit	Abbreviation
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	s	second (or hour)	sec. (or hr.)
Force	F	weight of 1 kilogram	kg	weight of 1 pound	lb.
Power	P	horsepower (metric)		horsepower	hp.
Speed	V	kilometers per hour	k.p.h.	miles per hour	m.p.h.
		meters per second	m.p.s.	feet per second	f.p.s.

2. GENERAL SYMBOLS

W, Weight $=mg$

g, Standard acceleration of gravity $=9.80665$ m/s^2 or 32.1740 ft./sec.^{2}

m, Mass $=\frac{W}{g}$

I, Moment of inertia $=mk^2$. (Indicate axis of radius of gyration k by proper subscript.)

μ, Coefficient of viscosity

ν, Kinematic viscosity

ρ, Density (mass per unit volume)

Standard density of dry air, 0.12497 kg-m^{-4}-s^2 at 15° C. and 760 mm; or 0.002378 lb.-ft.$^{-4}$ sec.^{2}

Specific weight of "standard" air, 1.2255 kg/m^3 or 0.07651 lb./cu.ft.

3. AERODYNAMIC SYMBOLS

S, Area

S_w, Area of wing

G, Gap

b, Span

c, Chord

$\frac{b^2}{S}$, Aspect ratio

V, True air speed

q, Dynamic pressure $=\frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L=\frac{L}{qS}$

D, Drag, absolute coefficient $C_D=\frac{D}{qS}$

D_o, Profile drag, absolute coefficient $C_{D_o}=\frac{D_o}{qS}$

D_i, Induced drag, absolute coefficient $C_{D_i}=\frac{D_i}{qS}$

D_p, Parasite drag, absolute coefficient $C_{D_p}=\frac{D_p}{qS}$

C, Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

R, Resultant force

i_w, Angle of setting of wings (relative to thrust line)

i_t, Angle of stabilizer setting (relative to thrust line)

Q, Resultant moment

Ω, Resultant angular velocity

$\rho\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension (e.g., for a model airfoil 3 in. chord, 100 m.p.h. normal pressure at 15° C., the corresponding number is 234,000; or for a model of 10 cm chord, 40 m.p.s. the corresponding number is 274,000)

C_p, Center-of-pressure coefficient (ratio of distance of $c.p.$ from leading edge to chord length)

α, Angle of attack

ϵ, Angle of downwash

α_o, Angle of attack, infinite aspect ratio

α_i, Angle of attack, induced

α_a, Angle of attack, absolute (measured from zero-lift position)

γ, Flight-path angle

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SUMMARY

A study was made to determine the air-flow characteristics around finned cylinders. Air-flow distribution is given for a smooth cylinder, for a finned cylinder having several fin spacings and fin widths, and for a cylinder with several types of baffle with various entrance and exit shapes.

The results of these tests show: That flow characteristics around a cylinder are not so critical to changes in fin width as they are to fin spacing; that the entrance of the baffle has a marked influence on its efficiency; that properly designed baffles increase the air flow over the rear of the cylinder; and that these tests check those of heat-transfer tests in the choice of the best baffle.

INTRODUCTION

Several investigations of the flow of air over flat plates and around smooth cylinders have yielded valuable information on boundary-layer phenomena. No published results are available, however, of the interfin velocities of the air flow around a finned cylinder. The velocity distribution in the fin space determines the boundary-layer characteristics at a given position around the cylinder. The fact that the entire mechanism of cooling is not given by these measurements does not influence the conclusions to be drawn relative to fin spacing, fin width, and pressure drop around the cylinder.

The present investigation was made to determine the effect of changes in fin width, fin pitch, and cylinder diameter on the interfin velocity of a cylinder model. Baffles found to be the best in tests of electrically heated cylinders (reference 1) were also tested. Interfin velocities with and without baffles for each of several positions around the cylinder were measured for five tunnel air speeds from approximately 38 to 145 miles per hour. The velocities in the exit passage of the baffles were in most cases measured for the same tunnel air speeds. The method of measurement made it impossible to measure the air flow in the boundary layer; it was possible, however, to measure the velocity distribution throughout the space between adjacent fins.

In a study of the entire problem of engine cooling it is imperative that the complete picture of air flow about the individual cylinders and baffles be known, for obviously the study of air flow is a very effective means of learning the conditions that give the best cooling with the least drag. Separate studies of flows over flat plates and over cylinders might supposedly be sufficient to give a working picture of the desired phenomena. It is believed, however, that the fins on the cylinder create a mutual interference, so that conclusions drawn from tests other than those of the combination itself might be misleading. The acceleration of the air around the front part of the cylinder, the deceleration around the rear, and the effect of the cylinder fins in stimulating the formation of the boundary layer on the cylinder are known to exist but are difficult to visualize in their proper perspective except by measurement.

Temperature measurements must be supplemented by air-flow measurements to show why one condition is good and another poor. Temperature measurements alone show only the condition; air-flow measurements, correlated with temperature measurements, show the cause.

APPARATUS

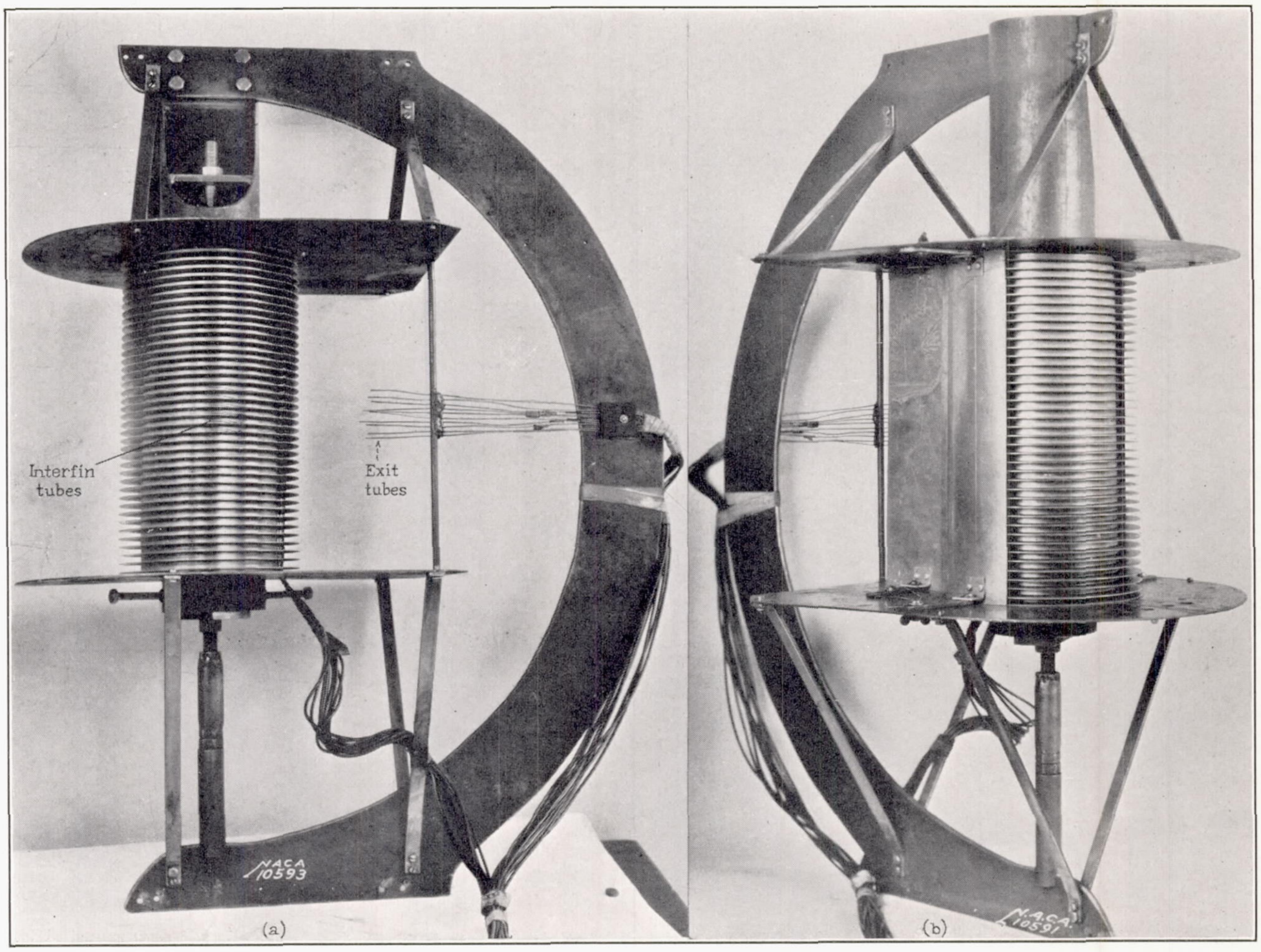
Wind tunnel.—The air-flow measurements around the cylinders were made in a 30-inch closed-throat wind tunnel designed to give air speeds up to 200 miles per hour. (See reference 2.) The tunnel air speeds were measured with a pitot-static tube located to one side and ahead of the test specimen to reduce the interference effect. A honeycomb grill in the tunnel entrance reduced air disturbances.

Apparatus for measuring the air flow around a cylinder.—The air flow between the fins and in the front and rear of the cylinder was in all cases measured with impact and static tubes. The tubes between the fins were placed tangential to the cylinder wall; the holes in the static tubes and open ends of the impact tubes were in a given plane passing through the axis of the cylinder. Never more than two tubes were located in a given space between two fins. An impact tube was located in one space and a static tube in the same position in an adjacent space, a procedure based upon the assumption that the flow through all the spaces used is identical. No justification can be given for such an assumption except that both static and impact pressures vary in a regular and reasonable manner from

cylinder wall to fin tip. Of the various means by which such a survey might have been made this method was the most practicable for such an extended survey.

The pitot-static tubes were made of stainless steel seamless tubing with a 0.002-inch thick wall. Two tube sizes were used; the 0.040-inch outside diameter tubes were used for the velocity survey ahead of the cylinder and in the baffle exit passages, and the 0.015-inch outside diameter tubes were installed between the fins to obtain the interfin velocities. The impact inch to the cylinder wall. Measurements across the fin spacing showed no measurable variation in velocity, indicating that the boundary layer along the fin was less than 0.010 inch. In the determination of the average velocity between the fins at a given position on the cylinder, the velocity was assumed to be constant across the fin spacing.

Cylinders and baffles.—The finned cylinders used in these tests were made of flat circular disks of different diameters clamped together with a ½-inch rod



(a) Interfin and exit tubes. (b) Baffles mounted in place.

FIGURE 1.—Assembly of finned test cylinder. The cylinder can be rotated about its central axis to obtain measurements at various angles with respect to the air stream.

tubes were made by cutting the tubes off square and removing the burs from the ends. The static tubes were made by closing and rounding off the ends and then drilling four 0.004-inch holes symmetrically around each tube about ¼ inch from the closed end. Although, as previously stated, never more than two tubes were located in a given space, all subsequent references to the tubes will be made as though they were all located in one space. Usually eight pairs of tubes were used to determine the air flow between the fins. Tubes were located both along the center line between the fins, and as close as 0.010 inch to the fins and 0.020 through their central axis. (See fig. 1 (a).) The fin disks were made of $\frac{1}{32}$-inch flat steel stock and the spacers of sheet aluminum. Two sets of disks were required to make up a cylinder, one set of large diameter serving as fins and another group of smaller diameter and the proper thickness forming the cylinder proper and giving the desired spacing between the fins. Both sets were always made from selected materials to secure perfectly flat surfaces and uniform thicknesses. The variation in fin spacing was never greater than 0.002 inch. Enough disks were assembled to give a cylinder 5 to 11 inches in length, depending on the

spacing. Cylinders of two base diameters, 4.66 and 7 inches, were used. The 4.66-inch cylinders had fin widths of 3/8, 3/4, 1 1/2, and 3 inches and fin spaces of 1/32, 1/16, 1/8, 1/4, and 1/2 inch. The cylinder could be rotated about its central axis to obtain readings at several points with one set of pitot-static tubes.

Baffles (figs. 1 (b) and 2) were made of 1/16-inch sheet aluminum annealed and rolled into shape. They were mounted around the cylinder symmetrically with respect to its central axis. One set of inner baffles (fig. 2, II–M) was made and tested to see if it were possible to guide the air farther around the rear of the cylinders. These baffles were partly slotted so that they could be installed between the fins.

TESTS

Air-flow measurements of the unbaffled cylinder combinations were made with air speeds of approximately 38, 56, 80, 110, and 145 miles per hour, at positions of $\theta = 15°$, 45°, 90°, 120°, and 135° from the front of the cylinder with fin spacings of 1/16, 1/8, 1/4, and 1/2 inch and fin widths of 3/8, 3/4, 1 1/2, and 3 inches. Tests of the 3-inch fins were limited to spacings of 1/8, 1/4, and 1/2 inch. The vibration of these fins made it necessary to use four sets of small interfin spacers located to offer a minimum of interference to the air flow in the vicinity of the pitot-static tubes. One set of tests was also made of the 3/4-inch fins using 1/32-inch spacing.

Both the velocity and the static pressure, read on a U-tube water manometer, were measured for all tests. The static pressure was used as a guide to detect errors or to determine any anomalous behavior. The manometer fluid showed no fluctuation because the small tubes damped the fluctuations in the air stream. The tubes checked a standard pitot-static tube to within 2 percent and the velocities are believed to be accurate to within ±5 percent.

Most of the baffle tests were conducted on the 4.66-inch cylinder having 3/4-inch fins with 1/4-inch spacers. The baffle used, found to be the best from previous tests (reference 1), fitted tightly against the fins and had an entrance angle of 140° and 3-inch extensions. The width of the exit passage was so proportioned that the ratio of exit area to area between fins at right angles to the direction of the air stream was 1.6, giving the optimum exit opening for the aforementioned cylinder a width of 2.1 inches.

Baffle I was a standard shell baffle with an entrance angle of 140° and an optimum exit 2.1 inches wide with 3-inch plates. (See fig. 2.) A complete velocity survey was made with the baffles in contact with and separated 1/2 inch from the fin tips. (See I_X and I_L, fig. 2.)

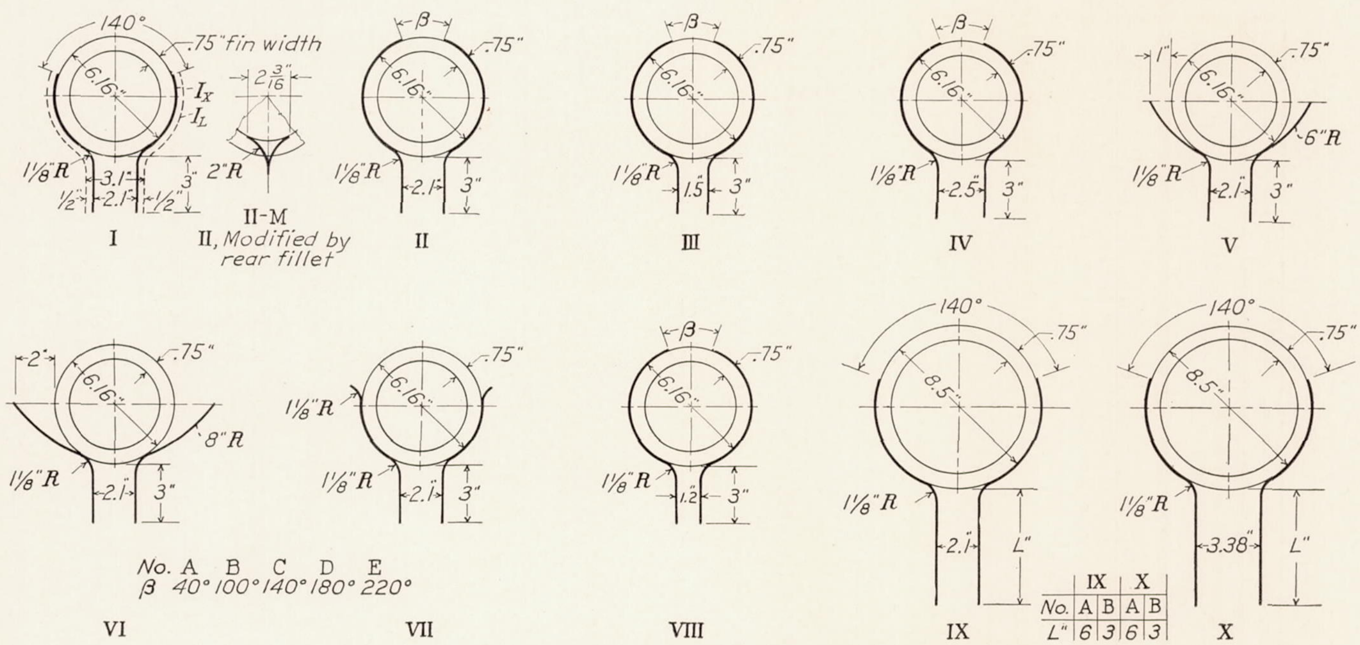


FIGURE 2.—Plan view of cylinder and baffle arrangements tested.

Complete velocity surveys with baffles II, III, and IV were made to determine the effect of changes in entrance angle and exit width on the flow characteristics around the cylinder and through the exit of the baffle. Five entrance angles (40°, 100°, 140°, 180°, and 220°) were tested with each of the three exit widths (1.5, 2.1, and 2.5 inches). The velocity survey for the baffle tests was made at angles of 15°, 45°, 90°, and 135° and, in addition, at points 5° ahead of and behind the forward edge of the baffle (e. g., 85° and 95° for the 180° entrance).

Baffles V and VI were tested to determine the effect of a pressure-type baffle on the flow characteristics around the cylinder. They had standard exit passages and flared entrances of 6- and 8-inch radii, respectively.

Baffle VII resembles II–D very closely and shows the effect of a small flare on the flow characteristics when used with a closely fitting baffle.

Baffle VIII was tested on the 4.66-inch diameter cylinder with 3/4-inch fins and 3/32-inch spacers. Velocity surveys were taken around the fins and through the exit passage for entrance angles of 40°, 100°, 140°, 180°, and 220°. The optimum exit passage was used on this baffle so that the width of the exit opening for this fin spacing was 1.2 inches.

Baffles IX and X were tested on the 7-inch diameter cylinder with 3/4-inch fins and 1/4-inch spacers. Velocity surveys were made with each baffle having 6-inch and 3-inch exit-passage lengths to determine the effect of the cylinder diameter on the optimum exit passage. Baffle X had the same entrance opening as baffle IX but the exit opening, instead of being 1.6 times the cross-sectional area between the fins, had the same angular opening in the rear as baffle II, so that the exit width was 3.38 inches.

In addition to the tests of finned cylinders, a similar velocity survey was made on a smooth cylinder 4.66 inches in diameter to compare the results with theoretical calculations.

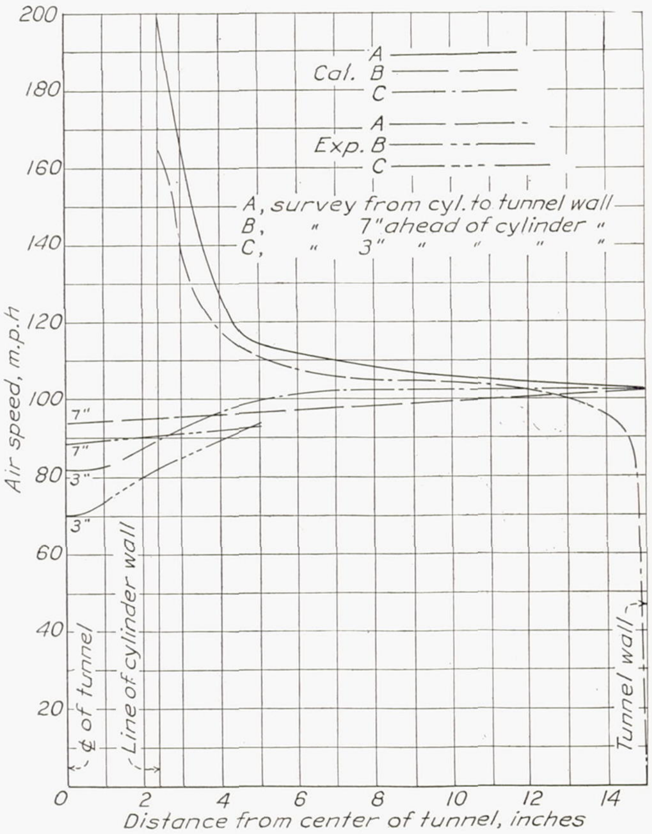


FIGURE 3.—Velocity survey of tunnel test section. V_t, 100 m. p. h.

RESULTS

Two velocity surveys were made in the throat of the tunnel at a tunnel air speed of 100 miles per hour; one along the diameter of the cylinder (extended) perpendicular to the axis of the tunnel, and the other at two stations ahead of the cylinder along horizontal lines perpendicular to the tunnel axis. The results of these surveys are shown in figure 3; the calculated curves are based upon a nonviscous potential flow in free space.

Although an integration of the experimental curves for air-speed distribution across the tunnel indicates that the pitot-static tube is so located that the readings are about 5 percent high, no correction has been applied because the uncorrected results will be more directly comparable with other studies made in the tunnel. Furthermore, the results presented here are more valuable for the comparison of flow characteristics under different conditions than for absolute values.

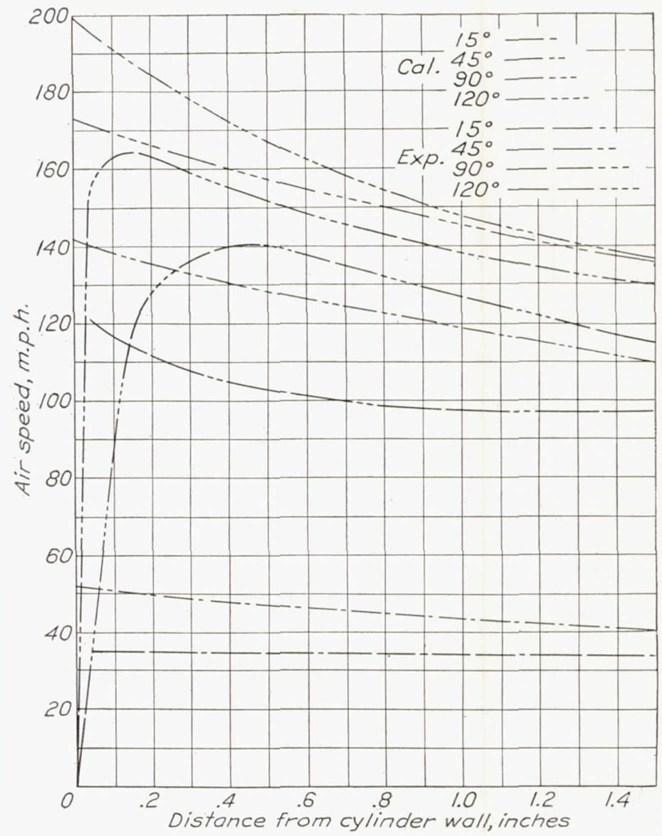
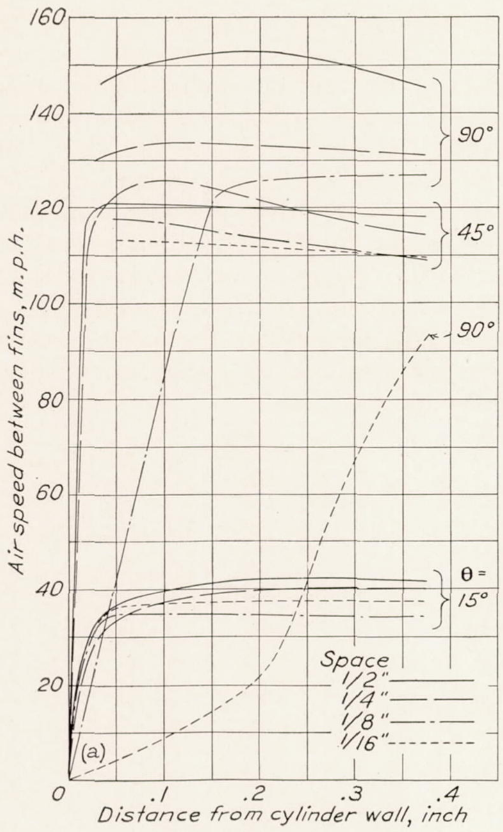


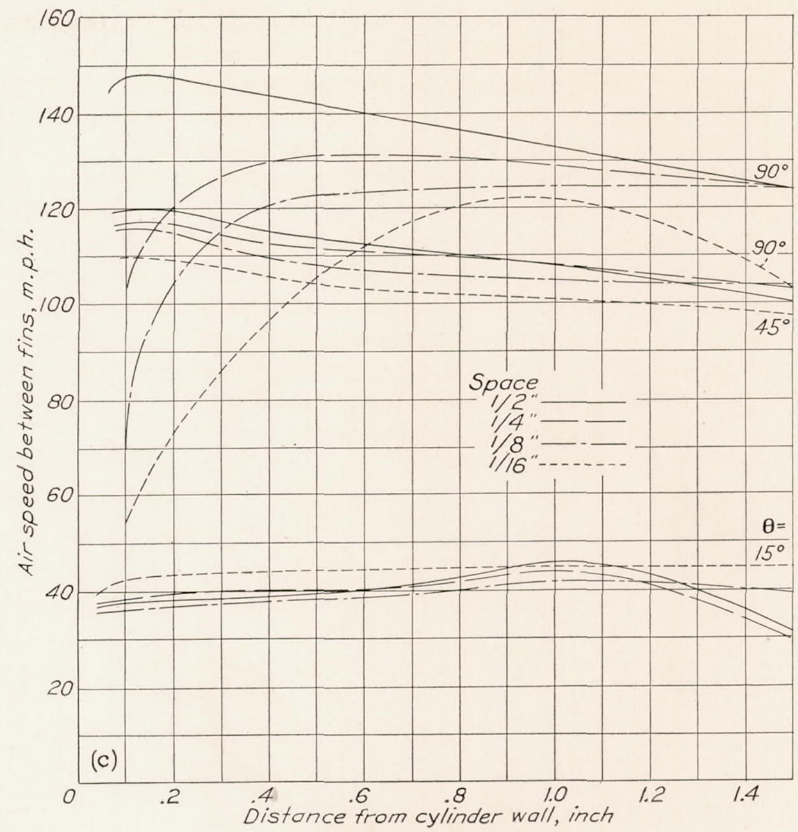
FIGURE 4.—Calculated and experimental air flow around a smooth cylinder of 4.66-inch diameter. V_t, 100 m. p. h.

Preliminary measurements were made of the air flow around a smooth cylinder. Figure 4 shows both the measured and the calculated air speeds. It is interesting to note that the experimental curves gradually approach the theoretical curves as the distance from the cylinder wall is increased and that the boundary-layer thickness for some of the settings is less than 0.040 inch, the closest point measured.

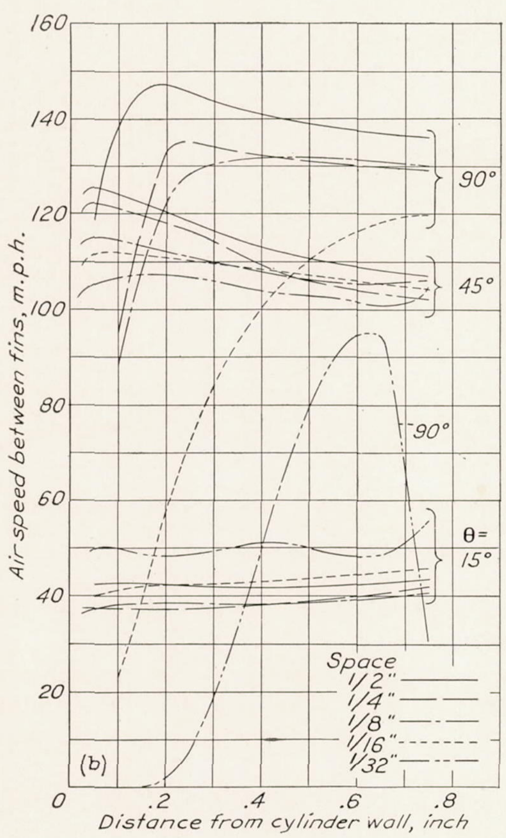
The data were reduced to values of V_a, average interfin velocity, and V_t, tunnel air speed, and the results were tabulated at 100 V_a/V_t. The data on the unbaffled cylinder are tabulated in table I; tables II to IV give the data for the complete series of baffle tests on three different cylinders, including the variations of baffle shape and of entrance and exit openings.



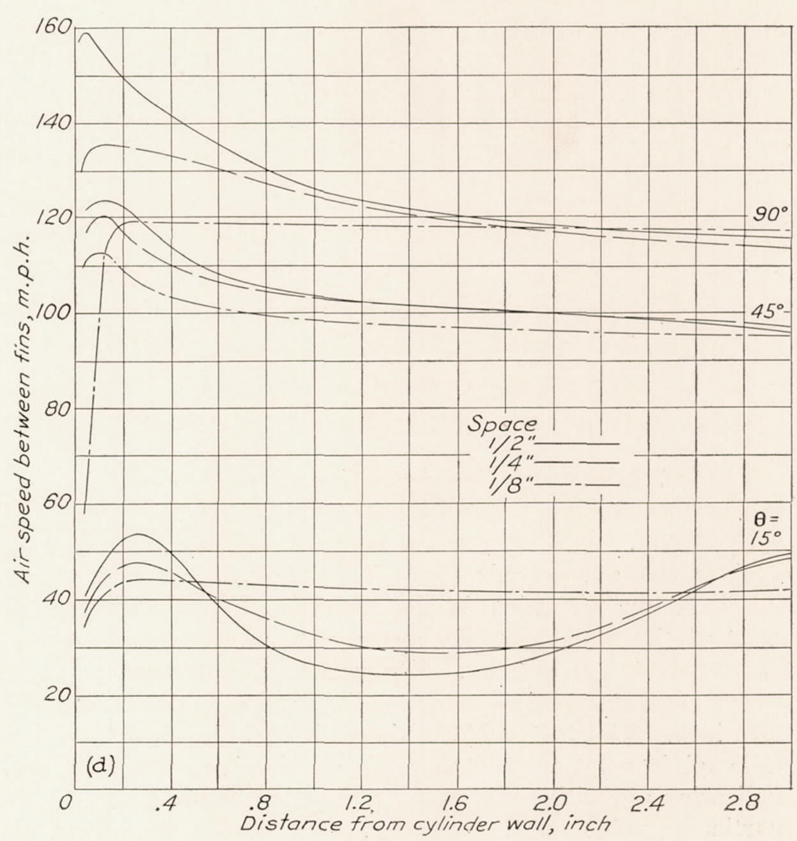
(a) Fin width, three-eighths inch.



(c) Fin width, 1½ inches.



(b) Fin width, three-fourths inch.



(d) Fin width, 3 inches.

FIGURE 5.—Effect of fin space on interfin velocity distribution of a finned cylinder. V_t, 100 m. p. h.

The results are given in two forms: As figures showing the air flow for typical cases, and as tables giving the ratio of average velocity (V_a) to the tunnel air speed (V_t) for the interfin and exit measurements. The plotted values are for a tunnel air speed of 100 miles per hour and the tabular data are for tunnel air speeds of 50, 100, and 150 miles per hour.

The results of tests of the unbaffled 4.66-inch diameter cylinder for all fin widths and spacings are shown in table I and figure 5. The fin spacing and the angular station (Θ) at which the measurements were made are designated on the figure. Each group of curves is for a particular fin width. Similar results for the 7-inch diameter cylinder are shown in table V and in figure 6.

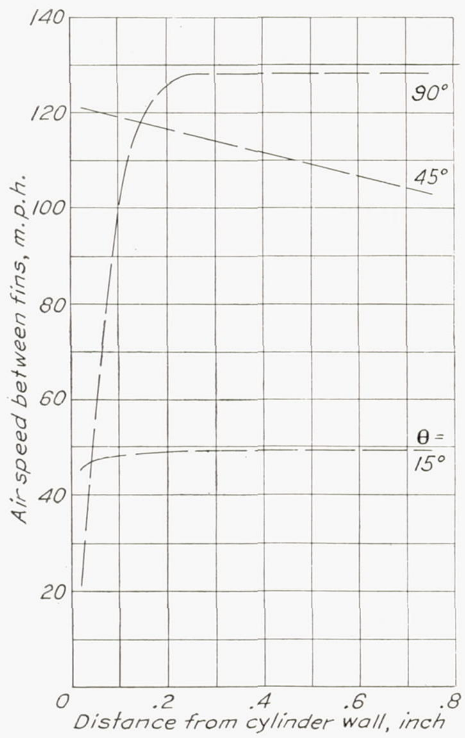


FIGURE 6.—Velocity distribution between fins of a 7-inch diameter cylinder. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.

Figure 7 helps to explain part of the phenomena encountered in the tests of cylinders with baffles. The curves are plots of Θ against the average air speed multiplied by a factor K, which is the ratio of the cooling area of the particular fin spacing to the cooling area of the ¼-inch spacing. The results presented are those for the 4.66-inch diameter cylinder with a ¾-inch fin width. The factor K multiplied by the air speed, although it is not an exact measure, should give a good indication of relative cooling for a given fin width at a particular point on the cylinder. The curves show definitely that better cooling should result from closely spaced fins. The best entrance angle (β) for a cylinder baffle for maximum cooling of the cylinder is given by the angle corresponding to the maximum value of KV_a.

All the test data for the 4.66-inch cylinder with ¼-inch fin spaces for various baffles are shown in figures 8 (a), 8 (b), 8 (c), and 9 and in tables II to IV. In figure 10 and in table VI similar results are given for the 1/32-inch spacing on the same cylinder diameter. Results of tests of the 7-inch cylinder are shown in figure 11 and table V. The results plotted in figures 8, 9, 10, and 11 agree with those shown in figure 7 in that the maximum air flow occurs with the baffle opening predicted by measurements on the unbaffled cylinder.

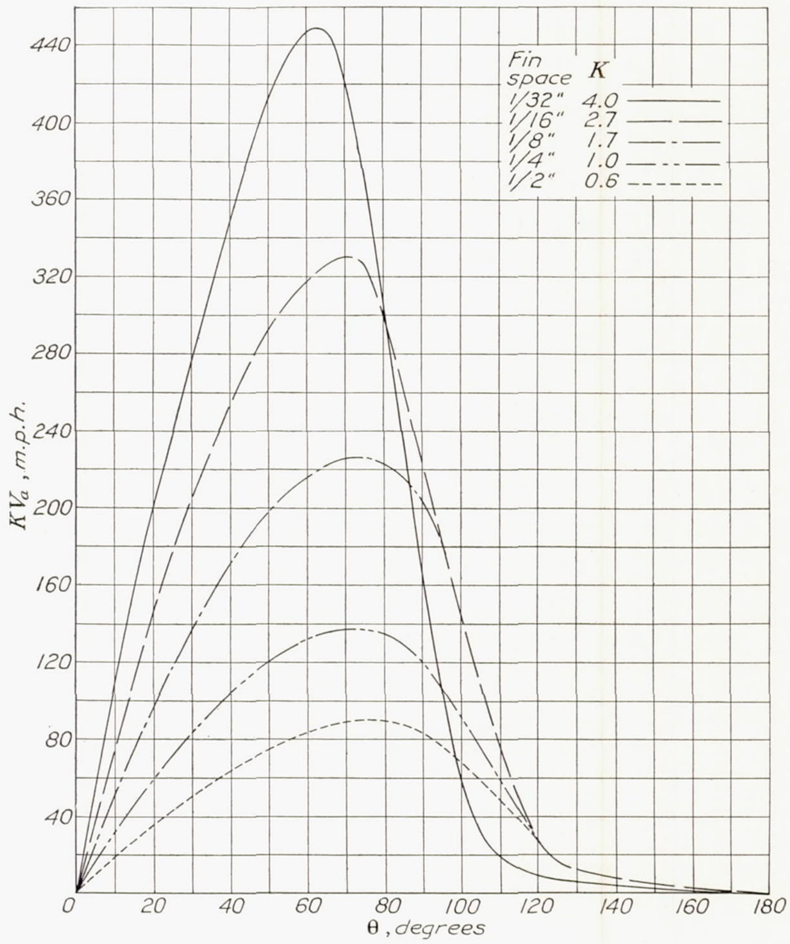


FIGURE 7.—Effect of fin space on cooling of a 4.66-inch diameter cylinder without baffles. V_t, 100 m. p. h. The best entrance angle for each fin space for a cylinder baffle for maximum cooling of a cylinder is given by the angle corresponding to a maximum value of KV_a.

The most desirable entrance angle, as shown by the figures, is 140°.

The results from tests of baffles II, III, and IV are shown in figure 12. The baffle with the 140° opening and 2.1-inch exit gave the maximum air flow between the fins.

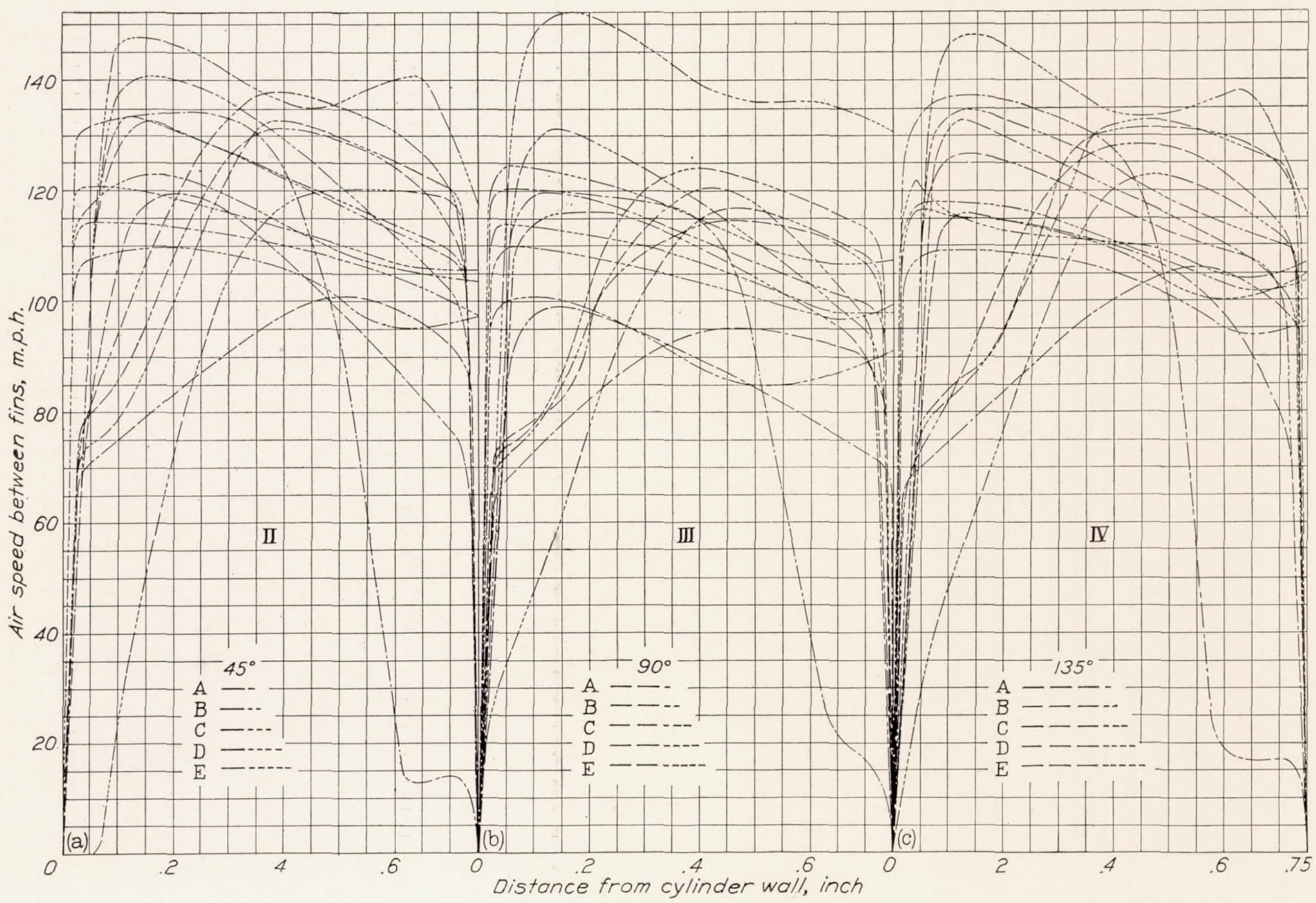
Figure 13 gives a direct comparison of the relative cooling of several of the arrangements tried in addition to the calculated and experimental velocity distribution around the 4.66-inch cylinder without baffles. Baffles II–C and VII are the best of the five plotted.

There are many published studies on boundary layer for various types of bodies. Doetsch, Éliás, Thom, Fage, and Pye are among those who have made experiments and calculations that are most directly applicable to this problem. (See references 3 to 10.) Doetsch studied the problem of velocity and temperature gradient near the surface of a heated flat plate in an air stream. His results check theory in that both gradient curves have the same form. The tests were run at low velocities so that the boundary layer was laminar. Thom has made boundary-layer experiments and calculations for the flow around a cylinder which are in close agreement but which, like those of Doetsch, are for laminar flow. Fage presents results, both experimental and calculated, for turbulent flow about a cylinder. Although the results given in the subject paper for a smooth cylinder at 45° (fig. 4) do not present satisfactory boundary-layer measurements, they do, however, agree with Fage's experiments at 49° in that the boundary-layer thickness is less than the closest point taken in these experiments. It is evident from the shape of the curves that the boundary layer is less than 0.040 inch thick. The boundary-layer thickness at 45° for a tunnel air speed of 100 miles per hour is calculated to be 0.035 inch. Pye has analyzed the entire mechanism of cooling around a finned cylinder. His article is a complete summary of the whole problem rather than a detailed study of any one part.

In figure 14 a comparison is made between a cooling and a velocity curve. The curves indicate how much cooler a given location is than the rear position on the cylinder wall at which the cooling is considered zero. This comparison, although not physically correct, is presented to correlate temperature and velocity measurements. The discrepancy is caused by the fact that the cooling is not shown to be due directly to boundary-layer conditions. The boundary layer and the velocities outside the boundary layer, however, are dependent upon each other, and therefore the comparison presented here has real physical significance even though it is slightly out of proportion.

The velocity corresponds to a tunnel air speed of 50 miles per hour. Two cases are presented: (a) No baffle, and (b), with the II–C baffle.

In both cases the cooling and velocity curves near the front of the cylinder are dissimilar, owing to the



(a) Exit width, 2.1 inches.

(b) Exit width, 1.5 inches.

(c) Exit width, 2.5 inches.

FIGURE 8.—Effect of exit width on interfin velocity of baffled cylinders having several entrance angles. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.

fact that an unmeasured radial component of velocity and a relatively thin boundary layer over the front of the cylinder exist and that relatively cool air comes directly in contact with the cooling surfaces. In view of these considerations, the lack of similarity is to be expected.

Farther around the cylinder the two cases must be treated separately. The expected similarity of the two curves of the unbaffled cylinder is evident. It would be incorrect, however, to ascribe the cooling over the rear two-thirds of the cylinder directly to velocity because the cooling-air temperature is rising, the boundary layer is getting thicker, and there is breakaway of the air flow from the rear that gives cooling phenomena rather easy to visualize but extremely difficult to predict. The vortices released at the breakaway certainly give excellent cooling in particular regions and undoubtedly keep the cylinder temperature in the rear much cooler than would be expected.

The baffled cylinder is even more illustrative than the unbaffled one because of the simpler phenomena. The velocity must remain the same; therefore, any change in the cooling must be due to changes of the boundary layer and the temperature of the cooling air. The velocity distributions shown in figure 8 indicate that the decrease in cooling is caused by the thick boundary layer next to the cylinder, emphasizing the importance of using closely fitting baffles.

Another factor of considerable importance in actual cooling is the type of cylinder baffle used, as shown in previous tests. The loss in total head as the air flows around a baffle-enclosed cylinder (fig. 15) is typical of all the combinations studied.

The curve for the 40° opening is obviously incorrect as energy once lost cannot be regained. It is believed that this erratic behavior is caused by the extreme turbulence at the entrance.

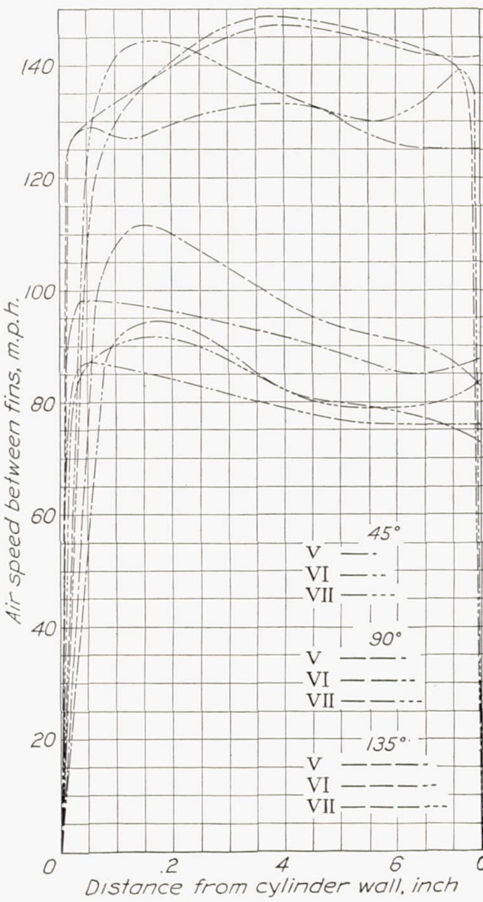


FIGURE 9.—Effect of baffle shape on interfin velocity. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.

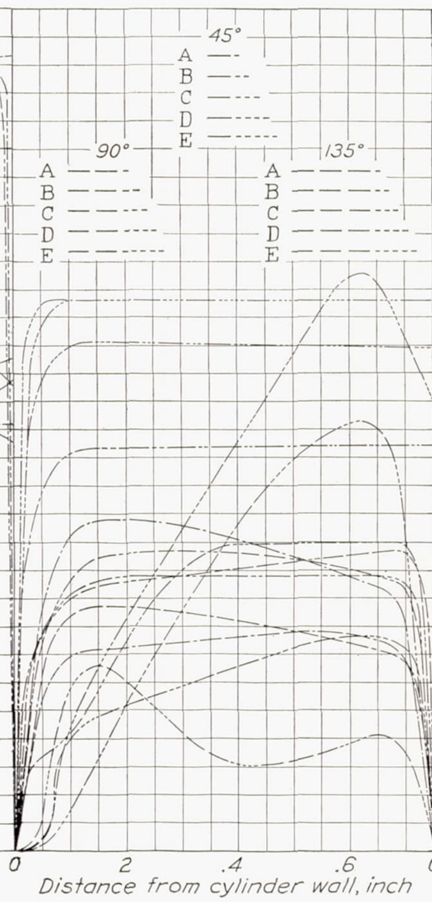


FIGURE 10.—Effect of entrance angle on interfin velocity. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one thirty-second inch.

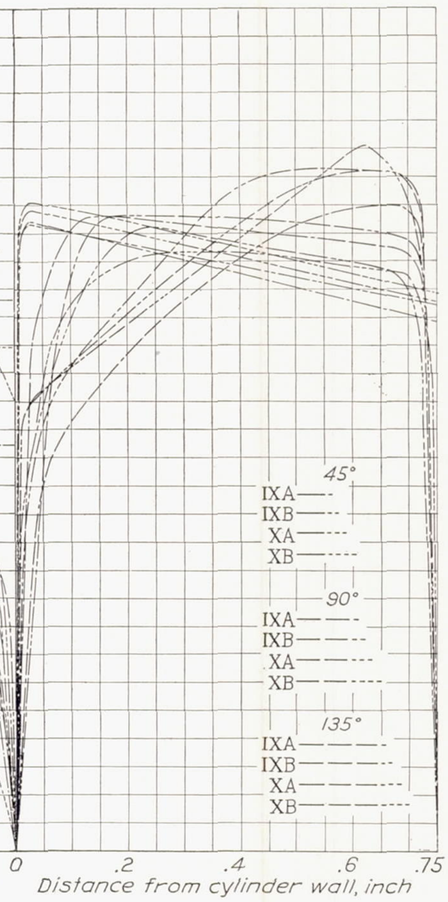


FIGURE 11.—Effect of exit width and length on interfin velocity. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.

The effect of the entrance angle β on the loss in total head is clearly shown in figure 15 for various positions θ around the cylinder. An entrance angle of 140° gives the most uniform loss in head and, as previously shown, provides the best cooling. It is apparent from these curves why one baffle is better than another.

Figure 16 shows the loss in total head through the baffles per inch of mean fin circumference for the 4.66-inch cylinder with ¼- and ⅟₃₂-inch spacings. Curve A is plotted for four sets of data, two exit widths and two entrance angles; and curve B is plotted for only one condition. The results show that the rate of pres-

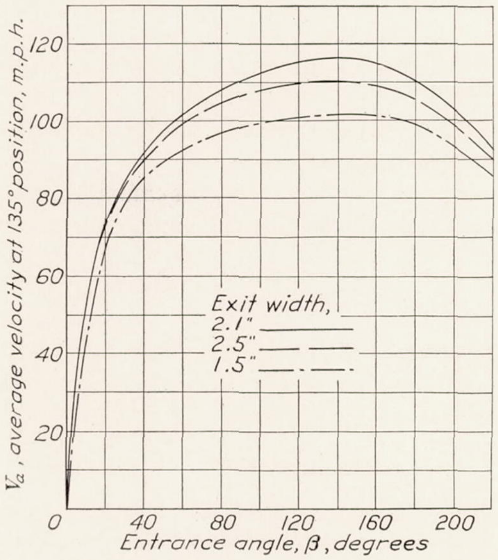


FIGURE 12.—Effect of entrance angle on V_a for a 4.66-inch diameter cylinder. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.

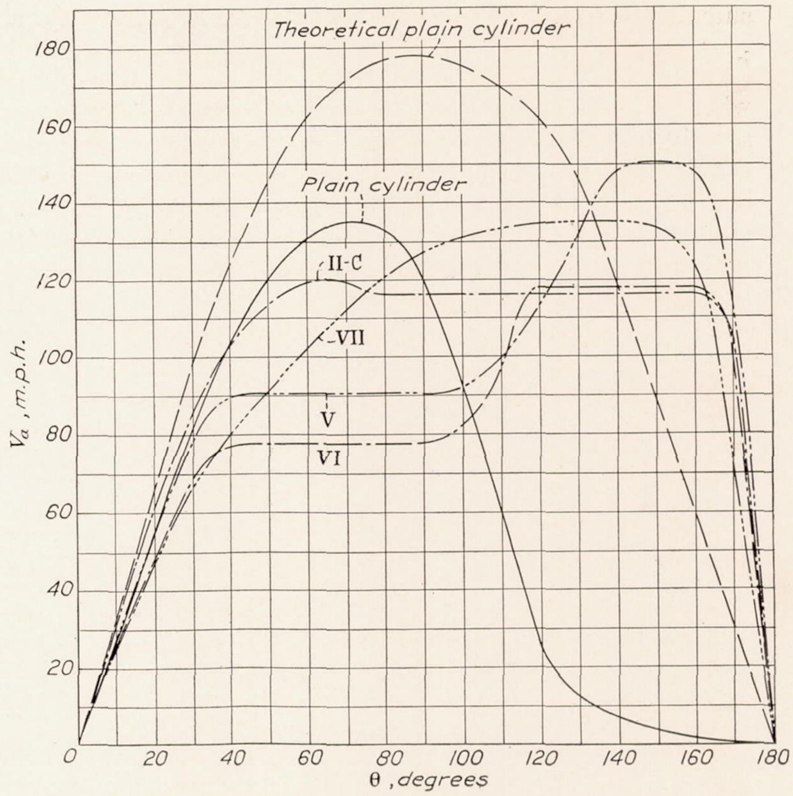
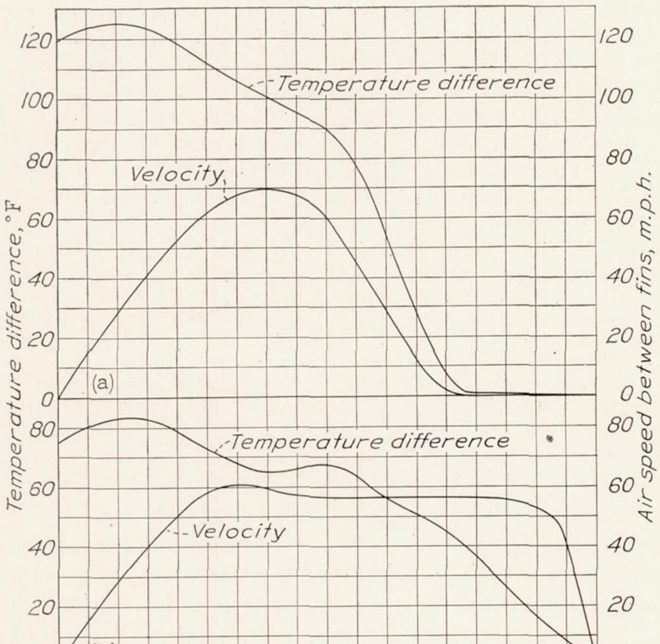


FIGURE 13.—Effect of design on the flow conditions of several baffles on a 4.66-inch diameter cylinder. V_t, 100 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch.



(a) Finned cylinder with no baffles. (b) Finned cylinder with baffle II–C.

FIGURE 14.—Comparison of temperature and velocity curves. V_t, 50 m. p. h.

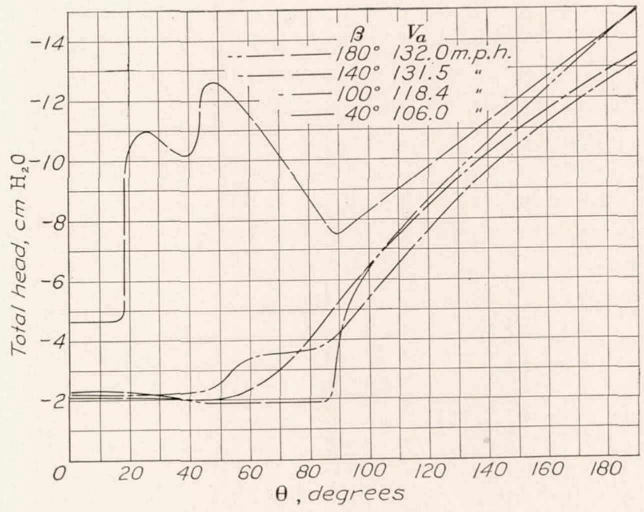


FIGURE 15.—Effect of entrance angle on the pressure drop around the cylinder with baffle II. V_t, 113 m. p. h.; fin width, three-fourths inch; fin space, one-fourth inch. The uniform loss in energy for the 140° opening results in good cooling.

sure drop through the baffle is very sensitive to changes in fin spacing. The rate of pressure drop $\frac{dP}{ds}=\alpha V^n$ where n is 1.8 and 1.6 for 1/32- and 1/4-inch spacing, respectively. Baffles giving a large loss in total head at the entrance or exit obviously cannot give a maximum interfin flow around the cylinders. Those having a uniform drop beginning just ahead of the baffle entrance gave the highest average air flow and over-all heat transfer.

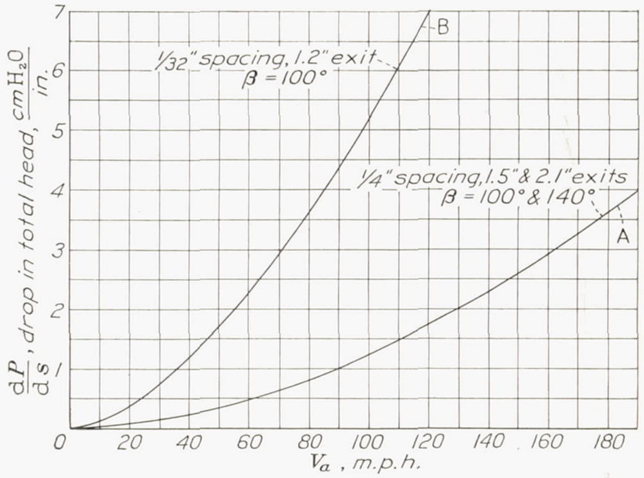


FIGURE 16.—Effect of fin spacing on differential pressure drop around the cylinder.

Some of the baffles in common use, termed "mixing" baffles, are placed in such a relation to the cylinder that they induce a mixing of the air at a point where more cooling is desired. As long as ample cooling is available, this practice is legitimate; but, when over-all cooling becomes a serious problem, such a wasteful practice cannot be continued.

CONCLUSIONS

1. The results presented here do not pretend to be directly applicable to solving the cooling problem on a cowled and baffled cylinder using *pressure cooling*. The results on pressure drop, the general behavior of the baffle in influencing the flow around the rear of the cylinder, and the study of fin spacing, however, are directly applicable. The study of entrance conditions for a baffled cylinder obviously applies only to a cylinder in a free air stream. Special emphasis should be placed on the results of the tests of flow around the rear of the cylinder, which show the very great importance of close baffling for maximum heat transfer and minimum energy loss.

2. Flow characteristics around a cylinder are not as critical to changes in fin width as they are to fin spacing.

3. Velocity measurements check temperature measurements in the choice of the best baffle.

4. The position of the entrance of the baffle has a marked influence on its efficiency.

5. The maximum air flow and the maximum heat transfer are both obtained with the same exit opening.

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *November 7, 1935.*

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TABLE I.—DATA FOR THE 4.66-INCH CYLINDER WITH NO BAFFLES

Fin width, inches / Fin space, inches	$\frac{V_a}{V_t}\times100$, where V_t=100 (50 m. p. h.; 150)				$\frac{V_a}{V_t}\times100$, where V_t=100 (50 m. p. h.; 150)			
	3/8	3/4	1 1/2	3	3/8	3/4	1 1/2	3
	Θ=15°				Θ=45°			
	34.1	42.0	41.6	----------	111.0	107.0	100.1	----------
1/16	36.2	42.5	40.2−	----------	111.5	108.3	100.4	----------
	58.0	43.3	*41.7*	----------	*114.0*	*110.4*	*107.2*	----------
	37.2	43.6	40.7	38.0	117.0	111.7	114.2	93.0
1/8	33.7	38.5	38.5−	41.3	113.7	108.4	104.2	98.3
	39.9	40.0	37.3	47.2	116.7	110.4	108.6	98.6
	37.2	42.9	42.1	34.7	118.7	114.2	106.6	99.2
1/4	38.0	38.7	40.0−	36.6	120.6	111.9	109.0	102.4
	41.9	45.3	38.8	40.5	*117.7*	*112.1*	*109.0*	*103.8*
	28.7	46.7	42.2	34.5	117.2	116.0	109.2	102.0
1/2	40.6	42.2	40.4−	35.1	120.0	114.8	110.8	103.2
	45.1	44.5	35.9	37.0	119.3	116.5	110.0	106.6
	Θ=90°				Θ=120°			
	18.5	63.2	103.2	----------	----------	19.0	21.7	----------
1/16	34.0	81.2	101.0	----------	----------	10.2	21.1	----------
	38.5	80.8	110.2	----------	----------	9.0	20.1	----------
	70.2	110.1	112.4	114.8	----------	20.1	42.0	52.4
1/8	101.2	116.2	114.8	115.6	----------	----------	56.5	59.4
	109.5	114.7	120.0	117.8	----------	15.8	57.4	62.0
	140.0	120.8	126.0	114.5	28.0	24.1	68.0	79.3
1/4	133.5	118.8	123.5	121.3	46.2	22.2	96.6	84.7
	123.3	120.5	124.7	123.5	54.2	27.1	91.0	89.0
	157.7	123.8	122.0	123.5	31.8	60.1	85.0	76.1
1/2	150.8	134.5	134.8	125.6	47.7	41.2	110.8	92.0
	153.3	141.7	139.3	127.2	109.0	43.3	109.5	91.3

TABLE II.—DATA FOR THE 4.66-INCH CYLINDER HAVING 3/4-INCH FINS AND 1/4-INCH SPACERS WITH BAFFLE I

Position / Θ degrees	$\frac{V_a}{V_t}\times100$, where V_t=100 (50 m. p. h.; 150)	
	I_X Baffle contacting fins	I_L Baffle 1/2 inch from fins
	42.2	49.6
15	49.0	53.7
	49.7	52.7
	110.0	96.8
45	108.6	107.0
	108.4	106.0
	120.2	127.2
65	119.3	127.3
	120.5	127.0
	115.6	104.8
75	116.6	129.5
	119.4	127.3
	113.4	115.8
90	116.0	118.5
	117.8	120.2
	113.6	116.6
135	117.8	126.6
	121.3	125.0
	117.2	106.2
150	116.6	114.1
	120.6	117.8
	76.0	94.6
Exit	77.1	86.9
	77.5	87.7

TABLE III.—DATA FOR 4.66-INCH CYLINDER HAVING ¾-INCH FINS AND ¼-INCH SPACERS WITH BAFFLES II, III, AND IV

$\frac{V_a}{V_t}\times 100$, where V_t = 50, 100, 150 m. p. h.

Θ degrees \ Exit width, inches	1.5	2.1	2.5	2.1 [1]
β=40°				
15	57.6	71.0	60.8	----------
	60.8	68.0	59.0	----------
	58.9	68.3	59.5	----------
25	81.6	93.6	90.4	----------
	82.0	89.5	92.6	----------
	85.3	95.8	94.6	----------
45	91.6	101.0	98.0	----------
	88.5	95.5	95.8	----------
	87.8	100.0	99.3	----------
90	89.4	101.2	95.8	98.4
	84.3	97.4	97.8	99.6
	90.6	98.7	99.3	98.0
135	90.8	94.6	89.0	----------
	84.4	89.2	91.0	----------
	83.0	89.5	91.7	----------
Exit	70.4	52.3	45.6	50.2
	63.8	52.2	48.9	46.0
	66.7	52.7	50.8	47.6
β=140°				
15	45.6	42.2	46.4	----------
	43.3	49.0	46.6	----------
	45.2	49.7	46.5	----------
45	100.0	110.0	110.4	----------
	102.0	108.6	107.9	----------
	103.6	108.4	109.9	----------
65	111.2	120.2	124.0	----------
	113.0	119.3	126.3	----------
	113.2	120.5	127.4	----------
75	99.6	115.6	115.8	----------
	105.3	116.6	117.9	----------
	105.7	119.4	120.5	----------
90	96.4	113.4	112.2	----------
	100.0	116.0	118.0	----------
	101.1	117.8	117.8	----------
135	96.4	113.6	107.0	----------
	98.4	117.8	110.2	----------
	101.1	121.3	111.7	----------
150	----------	117.2	----------	----------
	----------	116.6	----------	----------
	----------	120.6	----------	----------
Exit	91.0	76.0	71.0	----------
	94.2	77.1	68.8	----------
	95.7	77.5	67.3	----------
β=220°				
15	48.0	52.4	46.6	
	47.0	51.2	47.1	
	47.9	51.8	46.3	
45	112.4	113.6	108.0	
	113.8	113.7	109.8	
	115.3	116.0	108.9	
90	132.6	131.0	126.0	
	136.4	133.0	131.4	
	137.3	132.0	132.7	
105	102.4	104.0	101.0	
	114.0	107.6	107.2	
	114.0	107.0	106.1	
115	82.6	90.0	81.6	
	88.4	90.5	88.5	
	91.2	92.8	91.8	
135	72.4	75.0	78.6	
	90.7	89.4	92.6	
	94.8	91.3	94.7	
Exit	73.4	63.0	51.6	
	82.4	61.7	52.2	
	80.0	59.9	54.5	

$\frac{V_a}{V_t}\times 100$, where V_t = 50, 100, 150 m. p. h.

Θ degrees	1.5	2.1	2.5
β=100°			
15	45.2	45.2	45.4
	46.0	47.8	45.0
	49.0	48.6	47.0
45	92.0	97.8	96.4
	91.2	102.3	103.0
	90.2	101.8	103.0
55	100.0	105.8	104.2
	96.9	111.3	108.0
	101.9	111.7	110.7
90	100.0	106.6	103.2
	105.2	114.0	109.0
	102.9	117.9	112.1
135	93.2	104.4	103.4
	99.5	111.5	109.7
	104.8	116.0	111.5
Exit	84.4	69.4	53.2
	92.6	76.3	57.0
	94.2	77.5	60.3
β=180°			
15	43.4	45.0	42.4
	41.8	47.6	44.1
	43.0	48.8	43.6
45	104.6	109.8	105.4
	105.3	112.0	107.1
	105.3	113.4	110.0
85	112.4	121.0	117.4
	117.2	131.0	122.5
	118.6	131.3	123.4
90	104.8	115.0	105.6
	109.2	119.2	111.5
	110.2	121.3	113.3
95	101.2	111.2	103.8
	108.7	117.1	109.1
	109.5	120.4	111.2
135	100.2	105.2	104.6
	105.7	111.3	109.5
	107.0	115.2	113.2
Exit	90.4	73.4	57.2
	101.0	76.6	64.4
	101.0	79.7	63.1

[1] II, modified by rear fillet.

TABLE IV.—DATA FOR THE 4.66-INCH CYLINDER HAVING ¾-INCH FINS AND ¼-INCH SPACERS WITH BAFFLES V, VI, AND VII

Θ degrees \ Baffle	$\frac{V_a}{V_t}\times 100$, where V_t = 50 m. p. h., 100, 150: V	VI	VII
15	41.2	36.8	39.4
	41.2	36.4	38.2
	----	----	38.0
45	89.4	77.0	83.2
	91.4	78.8	83.2
	----	----	85.3
85	----	----	118.0
	----	----	125.0
	----	----	123.3
90	86.4	74.6	125.0
	94.2	79.7	132.0
	----	----	132.0
120	102.8	86.2	125.4
	117.7	98.0	135.2
	117.3	----	133.2
135	133.8	108.6	133.8
	140.1	128.2	136.5
	141.5	----	138.4
Exit	98.0	102.4	100.6
	106.2	134.5	9.75
	111.3	----	94.9

TABLE V.—DATA FOR 7.0-INCH CYLINDER HAVING ¾-INCH FINS AND ¼-INCH SPACERS

Θ degrees	$\frac{V_a}{V_t}\times 100$, where V_t = 50 m. p. h., 100, 150: No baffle	Baffle IX		Baffle X	
		A	B	A	B
15	50.0	43.8	40.6	45.6	47.2
	49.0	43.7	44.9	46.7	46.0
	48.7	44.0	44.9	46.7	46.5
45	111.4	103.2	102.8	104.6	106.4
	111.6	102.7	104.8	107.2	108.3
	113.3	103.0	104.2	106.6	108.0
65	----	112.0	112.6	112.6	118.6
	----	115.0	115.2	121.1	120.3
	----	115.7	115.0	123.3	120.3
75	136.2	100.4	104.0	96.4	95.4
	139.0	104.6	108.5	107.4	108.6
	139.3	107.0	111.7	111.3	111.0
90	103.8	93.8	99.0	91.4	87.8
	116.9	102.2	108.3	100.0	99.4
	116.6	106.5	109.0	103.3	102.7
135	----	90.2	100.4	99.6	98.0
	----	96.3	105.9	104.4	104.0
	----	99.6	109.0	108.0	104.6
Exit	----	69.2	67.2	41.4	46.4
	----	75.8	69.5	42.5	43.1
	----	77.6	75.5	45.4	45.6

TABLE VI.—DATA FOR 4.66-INCH CYLINDER HAVING ¾-INCH FINS AND 1/32-INCH SPACERS WITH NO BAFFLE AND WITH BAFFLE VIII

Θ degrees	$\frac{V_a}{V_t}\times 100$, where V_t = 50 m. p. h., 100, 150: β degrees	40	100	140	180	220
	No baffle	Baffle VIII				
15	43.6	36.2	33.6	37.2	38.2	39.2
	49.8	38.0	37.9	37.1	43.0	45.2
	52.7	38.1	38.6	38.6	45.2	45.7
25	----	50.4	----	----	----	----
	----	53.4	----	----	----	----
	----	50.0	----	----	----	----
45	88.2	46.6	60.0	74.6	78.6	86.0
	103.2	48.5	70.1	86.6	95.0	96.4
	104.6	50.1	71.3	87.5	97.3	100.3
55	----	----	45.2	----	----	----
	----	----	58.7	----	----	----
	----	----	59.6	----	----	----
65	----	----	----	55.6	----	----
	----	----	----	74.3	----	----
	----	----	----	82.0	----	----
75	71.2	----	----	37.8	----	----
	101.3	----	----	51.9	----	----
	104.7	----	----	56.4	----	----
85	----	----	----	----	45.0	----
	----	----	----	----	60.4	----
	----	----	----	----	61.3	----
90	29.4	36.8	38.4	33.6	27.0	37.8
	40.4	35.4	45.5	43.1	44.0	62.1
	41.4	36.8	49.5	45.8	44.9	63.9
95	----	----	----	----	25.6	----
	----	----	----	----	31.4	----
	----	----	----	----	36.0	----
105	----	----	----	----	----	13.0
	----	----	----	----	----	12.3
	----	----	----	----	----	12.9
115	----	----	----	----	----	----
	----	----	----	----	----	----
	----	----	----	----	----	----
135	----	32.6	36.2	34.6	21.2	20.0
	----	38.2	47.6	47.1	27.6	18.8
	----	43.1	56.1	49.3	33.8	19.6
150	----	----	----	----	----	18.4
	----	----	----	----	----	15.4
	----	----	----	----	----	16.6
Exit	----	41.0	----	----	----	----
	----	32.4	----	----	----	----
	----	38.1	----	----	----	----

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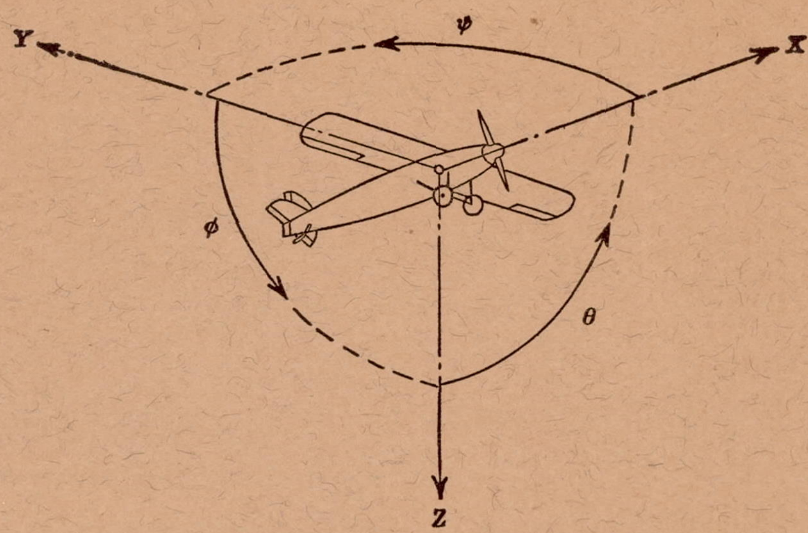

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	Rolling	L	$Y \longrightarrow Z$	Roll	ϕ	u	p
Lateral	Y	Y	Pitching	M	$Z \longrightarrow X$	Pitch	θ	v	q
Normal	Z	Z	Yawing	N	$X \longrightarrow Y$	Yaw	ψ	w	r

Absolute coefficients of moment

$C_l = \frac{L}{qbS}$ (rolling) $C_m = \frac{M}{qcS}$ (pitching) $C_n = \frac{N}{qbS}$ (yawing)

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter
p, Geometric pitch
p/D, Pitch ratio
V', Inflow velocity
V_s, Slipstream velocity
T, Thrust, absolute coefficient $C_T = \frac{T}{\rho n^2 D^4}$
Q, Torque, absolute coefficient $C_Q = \frac{Q}{\rho n^2 D^5}$

P, Power, absolute coefficient $C_P = \frac{P}{\rho n^3 D^5}$
C_s, Speed-power coefficient $= \sqrt[5]{\frac{\rho V^5}{P n^2}}$
η, Efficiency
n, Revolutions per second, r.p.s.
Φ, Effective helix angle $= \tan^{-1}\left(\frac{V}{2\pi rn}\right)$

5. NUMERICAL RELATIONS

1 hp. = 76.04 kg-m/s = 550 ft-lb./sec.
1 metric horsepower = 1.0132 hp.
1 m.p.h. = 0.4470 m.p.s.
1 m.p.s. = 2.2369 m.p.h

1 lb. = 0.4536 kg.
1 kg = 2.2046 lb.
1 mi. = 1,609.35 m = 5,280 ft.
1 m = 3.2808 ft.